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THE FUNDAMENTAL PRINCIPLES OF HIGH-SPEED SEMI-DIESEL ENGINES

By Dr. Büchner

PART III

A Discussion of Fuel Mixing and Ignition, with Special Reference to Engines with Precombustion Chambers

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In connection with Hesselman's experiments, it was incidentally shown how quantitative observations of the air currents, methodically made, may not only contribute to ultimate success, but may also form the indispensable basis for it. The Deutz horizontal solid-injection engine is a further proof of this contention.

It is one of the few solid-injection Diesel engines which antedate the war (Fig. 26). It works with a relatively low compression (25-30 atm.). When the fuel is gas oil, spontaneous ignition occurs at 25 atmospheres. Gas oil is therefore generally used as the priming oil, when a difficultly ignitible fuel, such as alcohol or coal-tar oil, is used for continuous operation. The engine is but slightly sensitive to the nature of the oil used.

* From "Jahrbuch der Brennkrafttechnischen Gesellschaft," Volume V (1924), pp. 90-106.

The engine has a divided compression chamber of the kind much used in small gas engines (Fig. 27). The larger compartment is formed by the cylinder proper, in which the piston works; the smaller, by the valve chamber into which open the fuel-injection valve, the air-intake valve and the exhaust valve. A cylindrical neck connects the valve chamber with the slightly conical end of the working cylinder. The working piston has a conical head (called "displacement head" or simply "displacer"), which, toward the end of the compression stroke, forms a constriction between the two chambers (Fig. 26) and, in common with the neck and the annular chamber, helps to form an annular opening and an adjacent, gradually expanding annular nozzle which, in turn, opens into the full circular cross section formed by the annular chamber.

By this diffuser, which has a variable cross section, there is generated, during the whole injection process, an enveloping mantle, which is in a state of lively agitation from its origin at the mouth of the annular nozzle, due to the frequent changes in direction and in cross section. This agitation is intensified when the individual air filaments, on issuing from the annular nozzle on their way toward the cylinder axis, mutually push one another from their direction of motion and assume the form of a hollow cone, which is reversed, as regards the cone formed by the annular nozzle, and opens toward the valve chamber and is gradually dispersed in eddies.

The completeness of the combustion, as proved by experiment, depends largely on the height of the "displacer" and the dimensions of the annular opening between the displacer and the cylindrical neck.

In every cylinder with its compression chamber divided according to Fig. 27, the piston works against two different compression ratios, $a - s : a$ for the annular space R and $b - s : b$ for the cylindrical center K. I am following K. Schmidt's explanation ("Zeitschrift des Vereines deutscher Ingenieure," 1922, p. 1125 ff). The pressure differences, which are consequently generated between the spaces R and K and which constantly strive to eliminate themselves, keep up a continuous flow from the space R toward the space K.

The curve e_1 in Fig. 27 represents, according to E. Schmidt, the kinetic energy, generated per unit of mass by the quantity of air passing from R to K per division of time or crank angle, as the piston approaches the dead center toward the end of the compression stroke, on the assumption that complete pressure equalization takes place between the spaces R and K. The course of the curve e_1 depends on the compression ratios determined by the dimensions of the engine and on the line of the mean compression pressure. With the ratios obtaining in practice, it reaches its maximum value at about 23° before the inner dead center, then falls rapidly in a concave curve, passes through zero at the dead center and then acquires negative values.

The curve e_2 applies to a 100 HP. engine with a "displacer." It falls less steeply between 20°and 2° before the dead center, so that the amount of energy required for improving the mixture and the combustion is available very near the dead center. Moreover, the energy available for the formation of eddies at the crank positions, corresponding to the beginning of the ignition, is about twice as great as in an engine without the displacer piston head.

After ignition, the mixing is assisted by the combustion of the fuel particles. Just after the dead center, as a consequence of the pressure difference between the spaces R and K, a retrograde motion sets in from K to R, which is likewise accompanied by strong eddies. Thus the air necessary for complete combustion is brought into contact with the still incompletely burned portions of the fuel charge.

In judging the "displacer" from the diagram in Fig. 27, it must be borne in mind that the special form of the air mantle determines the result. We can hardly expect any appreciable refinement of the atomization, since the region where the fuel and the air cone meet, is too far from the fuel and air valves, so that the velocities of the fuel and air particles are small at their crossing point. The fuel cone is received by the air mantle as by a shell and, since the latter is in especially lively agitation on its inner surface, the fuel clouds are dispersed, but are simultaneously confined in air and thus protected from immediate contact with the metal walls. The high air velocity in the cylindrical throttle opening produces a strong heat absorption, which is equivalent to an internal cooling of the valve chamber and a recovery of the fuel heat, which latter facilitates ignition.

The central fuel injection merits special emphasis. In the first development stage of the engine, the fuel was introduced laterally into the neck, so that it was dispersed by the air stream after the manner of the well-known "peeling" method of atomization. The transition to the central fuel injection resulted in a considerable saving of fuel.

It is frequently assumed that the Deutz displacement engine is possible only in the horizontal type, but the Deutz Engine Works, with its vertical VM engine, has made a solid-injection Diesel engine, whose combustion chamber has not the least similarity to their displacement engine.

That the displacement engine can be directly combined with the vertical type is demonstrated by the engine of Ruston and Hornsby, Fig. 28a (cf. "Engineer," of Feb. 15, 1924, p. 180), whose valve chamber, aside from the neck, has practically the same shape as that of the Deutz displacement engine. The stems of both inlet and exhaust valves are placed horizontally. The valves are combinations of disk valves with guide plates, which serve to relieve the valve spindles (mushroom valves). There is no "displacer" on the piston, but there is, between the valve chamber and the working cylinder, a neck with a noticeable constriction. In so far as this neck renders difficult the pressure equalization between the spaces before and behind it toward the end of the compression stroke, the air flow from the working cylinder toward the valve chamber will contribute to the agitation of the contents of the latter, up to the vicinity of the dead center, and, in the emptying of the valve chamber toward the working cylinder, the constriction helps to increase the homogeneity of the ignited fuel mixture.

The injection valve (Fig. 28b) corresponds exactly to the valve of Grieve-Livens shown in Fig. 9 (Part I). Since the valve chamber has a rectangular cross section with rounded corners, a flat fan-shaped fuel jet is produced, whose plane is perpendicular to the common axis of the inlet and exhaust valves. On the outer end of the valve stem, provided with labyrinth grooves, there is a second valve, whose seat forms a lift limit for the valve and allows the fuel escaping along the stem to enter a fuel pipe until it reaches its seat. This method of construction is intended to facilitate the working of the valve.

The method of improving the formation of the mixture and the combustion with the aid of a "displacer" piston has undergone many modifications. The engines of Tartrais-Peugeot, F. Ernst Bielefeld (Hamburg) and the Schneider Locomotive Works

(Winterthur) will serve as examples.

The Tartrais-Peugeot engine (Fig. 29), probably the first high-speed heavy-oil engine in Europe for motor cars, represents a transition phase between the hot-bulb engine and the solid-injection Diesel engine with spontaneous ignition. The hot bulb has the form of an annular chamber. It is closed at the top with a cover in the center of which the injection valve, surrounded by a water jacket, is located. The hot bulb is supported underneath by the cylinder head and secured by a threaded connecting tube, which is entered by the piston-head displacer near the end of the compression stroke. This leaves an annular opening for the passage of the compressed air from the main combustion chamber to the preliminary-combustion chamber. In the displacer itself there are small oblique channels which supply a weaker central air flow in conjunction with the annular air flow. The upper edge of the connecting tube is located slightly below the mouth of the fuel valve (Fig. 10a, Part I) and the top of the displacer, at the dead center, is only a little lower, as shown by the hatched outline of the displacer in Fig. 29. Hence the annular air stream strikes the fuel spray perpendicularly with great velocity, blows it to pieces and carries the fuel along with it into the hot bulb. According to Tartrais, the volatile constituents of the fuel evaporate immediately, while the heavier, still liquid, portions are hurled, by their centrifugal force, against the glowing walls of the hot bulb. Here they are also vaporized and the resulting vapors are carried along as indicated by the arrows. The fuel mixture is ignited by the combined action of the compression heat and the high temperature of the walls of the hot bulb. The central air stream is intended to hit the fuel valve at a lower velocity, so that the fuel will not be thrown against the relatively cool portions of the hot bulb in the immediate vicinity of the fuel opening. The constriction in the displacer is worthy of notice, since it produces (on the inner surface of the compressed annular air stream in contact with the fuel jet) a strong eddying motion which facilitates the mixing of the fuel and air and the combustion of the resulting product.

The following data on the type P.H.L.2 of the Tartrais-Peugeot engine are taken from a paper by I. L. Chaloner (Institution of Automobile Engineers, London, February, 1923, "High Speed Oil Engines").

It is a two-cylinder, two-stroke-cycle engine of 35-50 B.HP. of 120 mm (4.72 in.) cylinder bore and 150 mm (5.91 in.) piston stroke. Its revolution speed is 1400 R.P.M. and its weight per HP. is 5 kg (11 lb.). It has a special scavenging pump. Its good combustion is demonstrated by its low fuel consumption of about 180 g (6.35 oz.) per HP. at full load. The pressure at the end of the compression is about 20 kg/cm^2 (284.5 lb./sq.in.) and the maximum combustion pressure fluctuates between 28 and 32 kg/cm² (398.3 to 455.1 lb./sq.in.). The scavenging pressure is 0.35 kg/cm² (4.98 lb./sq.in.) and the injection pressure is about 45 kg/cm² (640 lb./sq.in.).

Table I gives the principal data of a twelve-day trial trip, for the purpose of testing the engine in omnibus traffic, regarding the utilizability of various easily obtainable fuels.

Table I.

Engine: Type P.H.L.2 (2-cylinder 120/150 mm).
Vehicle: Standard S.T.C.R.P. omnibus.
Total weight: 8128 kg (17919 lb.) (9 d. fully, 3 d. partly loaded).
Total distance: 1609 km (1000 miles).

Fuel	Specific gravity	Mean speed in km/hr.	Kilometers per liter	Fuel-cost ratio
Kerosene	0.805	19.0	2.9	1.00
Gas oil	0.835	17.7	3.1	0.35
Light fuel oil	0.885	17.1	3.4	0.23

On Oct. 2-3, 1922, a touring car was tried out on the route Paris-Bordeaux-Paris, with the results given in

Table II.

Engine: Type P.H.L.2 (2-cylinder 120/150 mm).
Vehicle: Standard 6-seat touring car (5 passengers).
Total weight: 2490 kg (5490 lb.).
Total distance: 1090 km (677 miles).

Fuel	Specific gravity	Speed in km/hr.		Kilometers per liter
		Mean	Maximum	
Gas oil	0.836	48.3	80.0	6.28

In order to compare this engine with a carburetor engine of the ordinary automobile type, two omnibuses of the same type, one equipped with a 4-cylinder carburetor engine (105/150 mm) and the other with a 35 HP 2-cylinder P.H.L.2 oil engine (120/150 mm), were driven two whole days, under perfectly similar conditions, with the results given in

Table III.

Engine	Fuel	Km per liter	Fuel-cost ratio
Carburetor (4 × 105 × 150)	Gasoline	2.3	1.00
P.H.L.2 (2 × 120 × 150)	Gas oil Light fuel oil	3.0 3.4	0.16 0.11

Table III shows a decided advantage for the P.H.L.2 engine over the ordinary carburetor engine, the fuel cost for the former, with light fuel oil, being only about 11% of what it was for the latter, with gasoline.

In the Bielefeld engine shown in Fig. 31, the precombustion chamber is pear-shaped and the fuel nozzle projects into it. Thus there is generated in this chamber an annular eddy which, on its inner side, strikes the fuel jet from above and carries the fuel vapor toward the "displacer." Bielefeld speaks of "air eddies in conjunction with fine atomization for producing quick combustion" and shows diagrammatically how the mixing can be advantageously accomplished.

The same principles are also applied in the engine shown in Fig. 30, which is likewise a two-stroke-cycle engine. The scavenging air is supposed to undergo a reversion of its direction of motion in the vicinity of the fuel valve, so that, near the end of the compression stroke, when the projection on the piston head approaches the projection on the cylinder head, an annular eddy is formed which cuts the plane of the fuel-valve opening (Fig. 30, below).

The two-stroke-cycle engine of the Schneider Locomotive Works at Winterthur (Fig. 32) also belongs to the type with divided combustion chamber and "displacer." The fuel is injected axially downward into the elongated, laterally-bulging precombustion chamber. The scavenging air likewise enters this chamber at the top through an annular valve. By means of spiral projections in the lower part of this chamber, the scavenging air is distributed over the larger cross section of the working cylinder and moves, in the direction of the cylinder axis, to the exhaust ports. Thereby the outer layers of air retain their circular motion. During the upward stroke of the piston, these layers (forced out of the working cylinder by the piston and retaining their rotary motion) again flow by the spiral surfaces and are deflected in the opposite direction, thereby being violently agitated and sometimes leaving an inner cylindrical column unaffected by the spiral projections and having essentially a simple axial flow. These violent eddies are still further strengthened when the "displacer" projection on the piston head enters the precombustion chamber. The displacer is relatively short and is rounded to fit the walls of the precombustion chamber.

The first German firm to put on the market a high-speed solid-injection Diesel engine is probably the "Hawa" ("Hannoverische Waggonfabrik"). The cylinder cross section of the latest polycylinder "Hawa" two-stroke-cycle crude-oil engine differs, however, from the form shown in Fig. 33, in that there has been added a special scavenging pump to take the place of the annular projection on the working piston which previously served to generate the scavenging air current. In other respects, the essential details of the original engine have been retained, the shape of the combustion chamber, the location of the exhaust valve at the upper end of the cylinder and the lateral position of the injection nozzle being especially noticeable.

The scavenging air enters the lower end of the cylinder through tangential slots. It is thus given a rotary motion, which is not only retained to the end of the compression stroke but is strengthened as a result of the displacer effect of the outer edge of the piston head and the decrease in the moment of inertia of the rotating mass of air. The fuel is injected into the lens-shaped rotating air disk in the form of a fan with the aid of a lip nozzle (last nozzle in Fig. 11, Part I), or (recently) in the form of a cone (Fig. 8b, Part I). The single-cylinder "Hawa" crude-oil engine furnishes 15 HP. at 600 R.P.M. The cylinder bore is 140 mm (5.51 in.). The piston stroke is 200 mm (7.87 in.). The fuel consumption is 200-220 g (7.05-7.76 oz.). The combustion air is compressed to 28-30 atm. (398-427 lb./sq.in.). No definite details have yet been published concerning the polycylinder engines, which are said to have been made in powers of 40-80 HP. with special scavenging pumps.

Articles have recently appeared in the newspapers on the Acro engine of the Acro Engine Company in Küssnacht (Switzerland) and the Dorner engine of the Dorner Engine Company in Hannover, which have attracted considerable attention. Regarding the Acro engine, we have at hand communications from Prof. A. Loschge (Munich) and Prof. Kurt Wiesinger (Zurich) from which the following data are taken.

It is a single-cylinder vertical four-stroke-cycle engine, furnishing 6 HP_e at 1000 R.P.M. and 9 HP_e at 1500 R.P.M. It has a cylinder bore d of 100 mm (3.94 in.), a piston stroke s of 140 mm (5.51 in.), a stroke volume V_h of 1.1 liters (1100 cm³ = 67.1 cu.in.) and a ratio $s/d = 1.4$. The length l of the connecting rod is 260 mm (10.24) in.), which gives a connecting-rod ratio $r/l = 1/3.72$. According to the statement of the firm, the compression ratio ϵ is 12, thus making the volume of the compression space 0.1 liter (100 cm³ = 6.1 cu.in.). With this compression ratio, the final compression pressure

70.7 BMEP

would be about 20 atm. (284 lb./sq.in.) at full load. The engine has two spring intake valves and two spring exhaust valves, which are actuated by an overhead camshaft.

The oil pressure in the pressure pipes is 80-100 atm. (1138-1422 lb./sq.in.) according to the statement of the firm. The regulation of the oil flow, to correspond to the load, is effected on the suction side by a device which was operated by hand during the experiments. There is also a device which renders it possible to change the injection point to correspond to the revolution speed and, lastly, a device for varying the tension of the spring in the nozzle.

The fuel was a dark mineral gas oil, with a petroleum odor, which had a specific gravity of 0.85 at 18°C (64.4°F). Its mean heating value was found by a Junkers calorimeter to be about 10,030 kcal/kg.

This gas oil has small quantities of volatile constituents, but consists principally of substances which distill only at high temperatures: 17% at 250°C (482°F); 55% at 300°C (572°F); 78% at 350°C (662°F). A considerable quantity of yellowish paraffin was deposited in the solid state in the condenser of the distilling apparatus. The residue in the distilling flask was dark and viscous and had a fatty-oily odor.

In fig. 34 the fuel consumption per HP./hr. is plotted against the effective horsepower at the four revolution speeds of n = 1000, 1250, 1380 and 1500 R.P.M. The revolution speed of the engine has comparatively little effect on the specific fuel consumption. For all revolution speeds from 1000 to 1500 R.P.M. it lies below 200 g .441# (7.05 oz.) from 5.5 to 10 HP_e and attains a minimum value of about 185 g .408# (6.53 oz.) per HP_e at 1380 R.P.M. and 7-8 HP_e. From this we calculate an efficiency of $\eta_w = \frac{632 \times 100}{10030 \times 0.185} = 34.1\%$, a result which has probably not been surpassed by any small high-speed engine.

The exhaust was invisible up to above normal load and contained no soot nor oil particles. It first became slightly colored under excessive loading. The exhaust temperatures were relatively low and reached values between 600 and 700°C (1112-1292°F) only at high loads and revolution speeds. The fuel-consumption figures at idling speed were always low. Prof. Wiesinger found a fuel consumption of 173.5 g .383# (6.12 oz.) for a minimum revolution speed of 500 R.P.M., with a lubricating-oil consumption of 8.7 g (0.31oz.) per hour. Prof. Loschge found a fuel consumption of 240-500 g (8.47-17.64 oz.) at a minimum idling speed of about 700 R.P.M.

In order to test the starting ability of the engine, cold water of 11°C (51.8°F) was run through the engine for about 10 minutes before cranking up. Even in this case, the engine was started. The pump was started by a few turns of the crank, so that it delivered the fuel to the injection nozzle in the right manner as soon as the engine was started, which never required more than one or two turns of the crank. It ran steadily, from the start, at its minimum idling speed.

Regarding the Dorner solid-injection 2-cylinder heavy-oil engine, Prof. Newmann of Hannover made the following statements. This four-stroke-cycle engine has two air-cooled cylinders of 700 mm (27.56 in.) 2.756 in. bore and 100 mm (3.94 in.) piston stroke, arranged in the form of a V. The normal revolution speed is 1400 R.P.M. The cylinders work with spontaneous ignition and solid injection produced by a fuel pump for each cylinder. The specific gravity of the gas oil was 0.864.

W.F.S.

Two experiments were tried (the first with normal load and the second with excess load) with the following results:

Experiments	I	II
R.P.M.	1400	1400
Brake weight	3.8 kg(8.378 lb.)	5 kg(11.023 lb.)
Fuel volume per hr.	1436 cm³ (87.6 cu.in.)	2084 cm³ (127.17 cu.in.)
Effective HP.	4.5	6
Fuel consumption per hr. per B.H.P.	1.24 kg(2.73 lb.)	1.8 kg(3.97 lb.)
Fuel consumption per HP_e/hr.	0.275 kg(0.606 lb.)	0.3 kg(0.661 lb.)
Mean eff. piston pressure	3.75 atm.(53 lb./sq.in.)	5 atm.(71 lb./sq.in.)
Thermal efficiency for 10000 kg-cal or, 17999 B.t.u./lb.	0.23	0.21

The combustion was perfect. After the conclusion of the experiments, the inside of a cylinder was inspected and found clean, both as regards the sides and the top of the combustion chamber. It follows from the mean effective piston pressure that the cylinder volume was well utilized, it being 3.75 atm. (53.3 lb./sq.in.) for 4.5 HP_e. The engine could be temporarily overloaded up to 6 HP_e, whereby the mean effective piston pressure rose to 5 atm. (71 lb./sq.in.). For an increase of 33 1/3% in the effective HP., the fuel consumption rose only 9%.

Due to the peculiar structure of the nozzles, they were not harmed by small particles of dirt in the fuel, since they are self-cleaning to a certain degree.

In order to render the atomization visible, the fuel pump and nozzle were unmounted and the gas oil was sprayed into the open air. It gave an extremely fine conical spray, which could be easily ignited with a match.

The Dorner Engine Company of Hannover added the following comparison of the operating cost of an automobile for two or three persons (or a delivery wagon for 250 kg (551.2 lb.) of useful load), when driven by a heavy-oil engine, with the operating cost of the same vehicle when driven by a gasoline engine.

1. Operating cost for the gasoline vehicle.

Gasoline	1.52 marks		
Lubricating oil	0.16	"	
Tires	1.43	"	
Total	3.11	"	for 100 km, or about 3.1 pfennigs per km.

2. Operating cost for the heavy-oil vehicle.

Gas oil	0.33 marks		
Lubricating oil	0.16	"	
Tires	1.15	"	
Total	1.64	"	for 100 km, or about 1 2/3 pfennigs per km.

The heavy-oil vehicle therefore shows a saving of nearly 50%.

However strongly the interest of the public has been aroused by the above information, it will naturally not be entirely satisfied. It is obvious, however, that, as regards the revolution speed and the fuel consumption, a turning point has been reached in the development of high-speed semi-Diesel engines.

Although the sources from which to draw information and inspiration in the field of high-speed half-Diesel engines, now flow very scantily and therefore every attempt to present their principles can only be regarded as an imperfect sketch, this does not preclude the possibility of finding, even now, the characteristic lines to which every future presentation of the subject must refer. To me the following seems especially important and therewith I will resume my previous line of thought.

Many processes in carburetor engines could be satisfactorily explained only after we had become accustomed to think of the currents and eddies arising in the combustion chambers. These contribute greatly to the improvement of the heterogeneous mixture before combustion and more than offset the harmful cooling effect of the metal walls on the burning gases by transmitting heat to the still unkindled mixture. The mixture can be rendered turbulent either during the suction stroke, by the high intake velocity and the deflecting action of the intake valve or, during the compression stroke, by the changes in the shape of the compression chamber and by the shifting of the center of gravity of the cylinder charge. The second method is especially efficacious when it takes effect immediately before the passage of the spark. Often this situation has automatically arisen when (in spite of the advocacy in technical literature of the so-called spherical combustion chambers) divided combustion chambers were employed, with the valves located in side chambers. Often, however, the advantages connected with turbulence are intentionally increased to the utmost, as in the "turbulence engines" in which there are constrictions between the working cylinder and the valve chamber.

The six combustion chambers in Fig. 35 are from the text book of H. A. Huebotter, M.E. ("Mechanics of the Gasoline Engine," pp. 26-27, New York, 1923). In four of the combustion chambers, arrows indicate the nature of the currents and eddies produced by the marginal and deflecting action of the intake valves. There are, moreover, two types of cylinders represented, in which (near the inflowing currents and located above them) the currents and eddies generated during the compression stroke are especially prominent, because the shape of the compression chamber changes suddenly, as the piston approaches the top dead center, and the mixture, driven out of the cylinder by the piston, must pass through narrow channels into the valve chamber or chambers.

Just as, in carburetor engines, both divided and undivided combustion chambers have been successfully used, so also, in solid-injection semi-Diesel engines (either low or high speed), no standard combustion chamber has yet been evolved, not even in the two main groups, which are customarily distinguished, namely, the precombustion engines and solid-injection engines with undivided combustion chambers.

Fig. 36 shows a few examples of precombustion engines in which the volume of the precombustion chamber is but a small fraction of the whole combustion space. These are the engines of the Worthington Pump and Machinery Corporation, U.S.A., of the "Western" Suter (type B.V.), U.S.A., E. T. Adams and Sons, Los Angeles, California, U.S.A., Mannheim Engine Works (formerly Benz and Company), stationary engines, Heinrich Kämper Engine

Works, Marienfelde, and the Deutz Engine Company. In the first three makes the precombustion chamber is strongly cooled. The "Western" Suter Company lays especial emphasis on the need of keeping the walls of the injection nozzle and precombustion chamber at a uniform temperature by cooling with water, in order that no change take place in the fuel and that no combustion residues be deposited on the walls. The engines of Benz and Company (Mannheim Engine Works) and of the Kämper Engine Works have an ignition lining in the precombustion chamber, which has only small contact surfaces in common with the cooled portions of this chamber and which consequently produces, in the portions touched by the fuel, the high temperatures required for ignition. In the Deutz engine, which in future will replace the Brons engine, the precombustion chamber is moderately cooled (cf. Prof. Nägel, "Dieselmaschinen," published by the "Vereines deutscher Ingenieure," Berlin, 1923, p. 28). The Deutz engine has, moreover, a second injection nozzle for introducing fuel directly into the disk-shaped main combustion chamber.

When the precombustion chambers are kept at moderate temperatures, the shape of the diffuser and of the immediately adjoining portion of the precombustion chamber requires special attention. Here differences which, regarded externally, may seem unimportant, can produce much altered flow relations and therefore decisively affect the mixing before the beginning of the combustion. The relatively low injection pressure is often represented as an especial advantage of precombustion engines. Pressures in the vicinity of 50 atm. (71.1 lb./sq.in.) are sometimes mentioned. Thereby it should not be forgotten that fuel and air often meet inside the diffuser, in pure counter-flow, under conditions where, as the result of the over-expansion of the air stream, air velocities are generated which exceed the velocity in the narrowest passage, namely, the neck of the diffuser. Fuel nozzle and diffuser together constitute an inseparable whole in judging their combined action.

Although but few of the details of the fuel nozzles of successful precombustion engines have yet been published, it may at least be said that the present tendency is to avoid spreading the spray excessively on its axial introduction into the precombustion chamber. Hence the preference is given to nozzles with annular openings and small lateral angles. The fear of "selective" combustion, i.e., of premature combustion of the volatile and easily ignitible constituents of the fuel may have contributed to this preference, as likewise the desire to retard the "collective" combustion (the combustion of the fuel as a whole) and, in conjunction with the pressure increase at the beginning of the combustion, to obtain as long a constant-pressure line as possible.

The engine of the Worthington Pump Machinery Corporation (Fig. 36) merits a brief explanation. It has slot-shaped scavenging ports, the intake ports being directed steeply upward. The degree of contact between the air and the injected fuel and, consequently, the degree of the combustion in the precombustion chamber are determined by the kind of fuel and by the direction and the surface area of the injected drops. The fuel cloud approaches the opening, which connects the precombustion chamber with the main combustion chamber, leaving a certain reserve supply of air in the outer part of the precombustion chamber. This reserve air is driven, along with the fuel cloud, out of the precombustion chamber into the main combustion chamber and mixes with the fuel, so that the combustion lasts longer, than if all the air were thoroughly mixed with the fuel at the outset. The diffuser neck consists of an easily replaceable piece of sheet metal, which has a conical shape on the precombustion end and either a cylindrical or conical shape on the cylinder end. The edges are sharp, so that it may be regarded as an opening in a thin wall.

In all the other engines represented in Fig. 36, a series of holes is employed for connecting the two combustion chambers, as is customary in ignition-cartridge engines ("Zundkapselmaschinen"). Where is the line to be drawn between ignition-cartridge and precombustion-chamber engines? Even in ignition-cartridge engines there has recently been an increasing tendency not to store up fuel in the cartridge during the intake stroke, but to inject it during the compression stroke.

The direction and number of injection holes are adapted to the shape of the main combustion chamber. The above-quoted observations (of Mackechnie, Deutz, Hesselman and others) give important indications concerning the introduction of round liquid jets into the combustion chambers, although a preliminary mixing produces a more elastic fluid than the liquid fuel.

No detailed experimental results are at hand on the flow processes in the precombustion chambers of precombustion engines. Perhaps the researches of the Semmler Consortium on gas turbines (cf. also Gentsch, "Untersuchungen der Gas- und Oelgleichdruckturbine"), can throw a little light on the subject, as also, conversely, the development of gas turbines may be promoted by our increased knowledge of the motion of burning gases from the observation of precombustion engines.

Furthermore, I call your attention to the cross-sectional development of the precombustion chamber in the direction of its axis, to its volumetric relations in comparison with the main combustion chamber, to the location of the injection opening inside the chamber and to similar, purely geometrical but decisive characteristics for the processes in the cylinder cross section according to Fig. 36.

In conclusion, I will refer to a small collection of combustion chambers which, in contrast with the precombustion engines, have a relatively closed form, including the combustion chambers of the Augsburg-Nuremberg Engine Works, of Krupp and of Professor Junkers, which are still regarded as standard in Germany. However much the outlines of these combustion chambers in Fig. 37 may differ from those in Fig. 36, there is still a common bond. Heraclitus says: "Everything flows." This applies to all fields of human endeavor, to small and great events. It also applies to that branch of mechanical engineering which forms the subject of my lecture. Sperry's engine (Fig. 37), with its high-pressure combustion chamber and the side chamber in which is located the valve leading to the mean-pressure cylinder, reminds us of the carburetor engine with the side chamber as shown in Fig. 35. Also in the remaining engines, strong air currents are generated at the end of the compression stroke, since we here have to do with graded compression chambers whose separate compression stages have distinguishable compression ratios.

Romeyn's engine has a special stationary "displacer" on the cylinder head, which generates a spiral eddy in the annular combustion chamber, the fuel entering the latter at one or more points.

Even in Junkers' combustion chamber, whose diagrammatic representation in Fig. 37 at first gives a purely static impression, there is much agitation, since violent eddies are produced in the fuel jet itself by the special nozzle and the jet enters an atmosphere which has not regained its equilibrium after the conclusion of the scavenging process.

I have now reached the end of my remarks. I have more than once realized that the data at my disposal are very scanty. They are discoveries upon which I stumbled as I turned my glance and steps afar, but the fact that they come from both home and abroad imparts to them perhaps some little value.

In spite of all striving for individual freedom, in spite of zeal for one's own original creations, the inventor must not lose sight of the accomplishments of others, lest he fall behind in the contest with individuals and nations. My desire is that this discussion may help to give my colleagues a broader view of the whole field.

Translation by Dwight M. Miner,
National Advisory Committee
for Aeronautics.

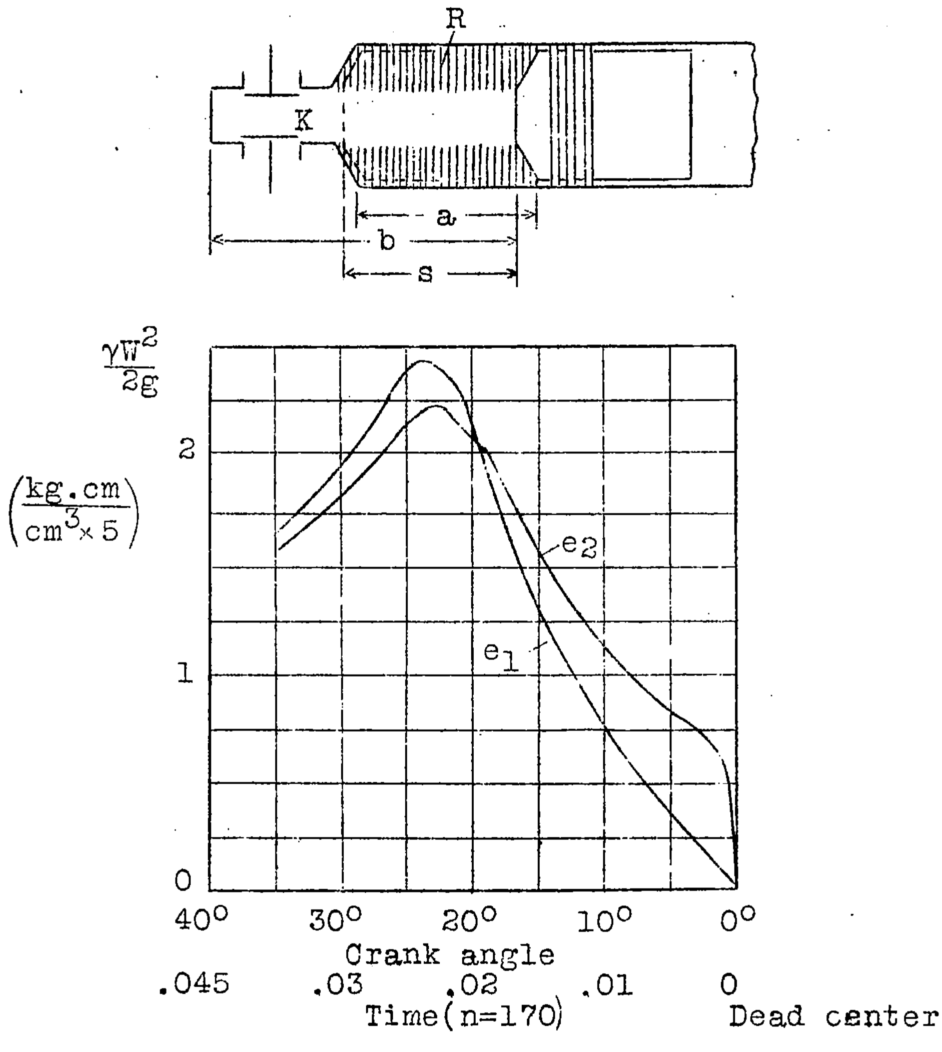

Fig.27 Diagram of combustion chamber showing different compressions.

Deutz
"Displacer" engine

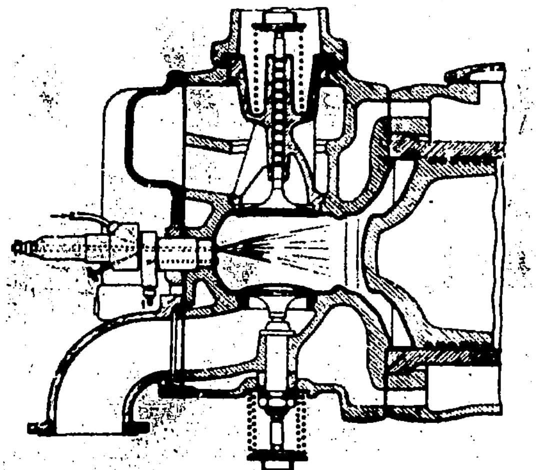

Fig.26

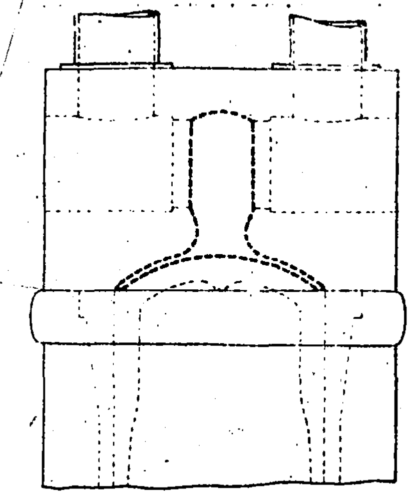

Fig.28a

Ruston and Hornsby

Tartrais - Peugeot

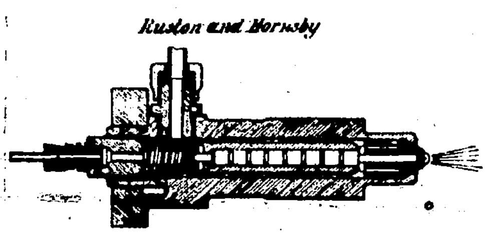

Fig.28b

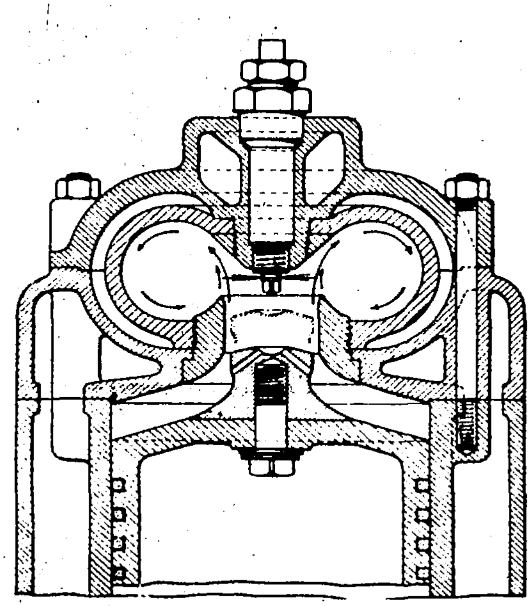

Fig.29

F. Ernst Bielefeld

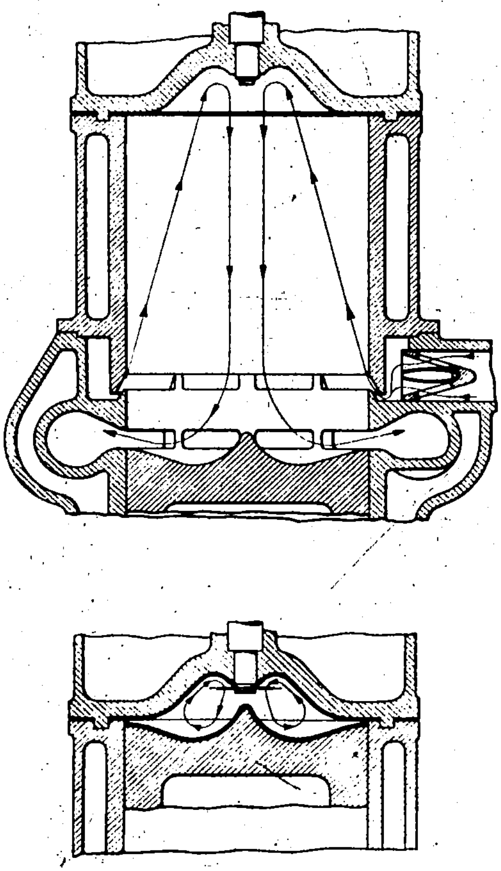

Fig.30

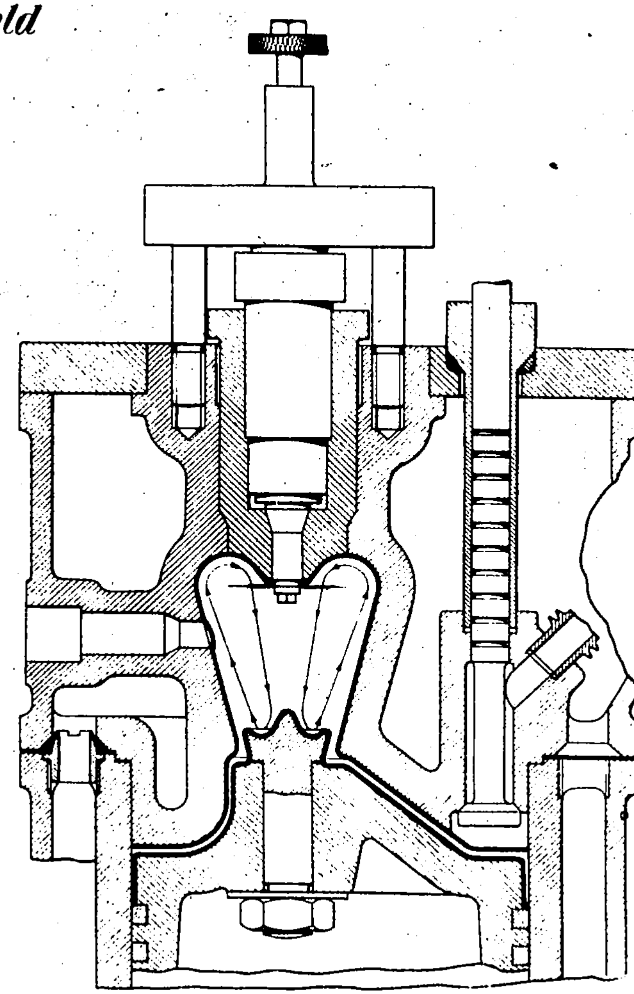

Fig.31

Schneider

"Hawa"

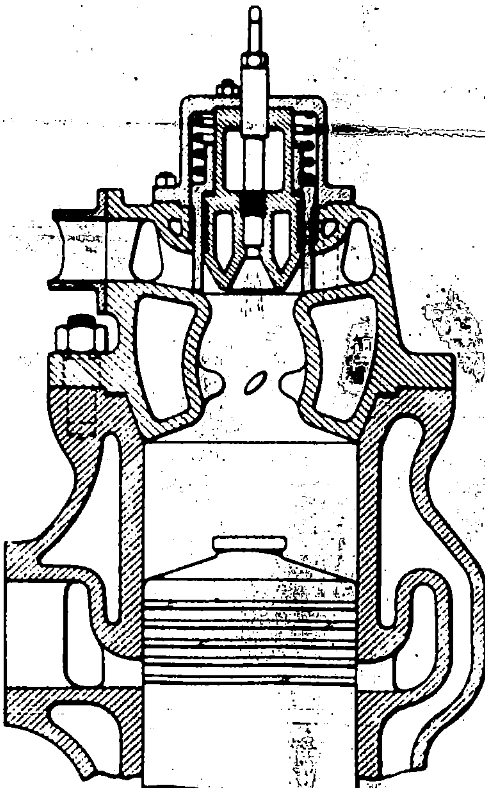

Fig.32

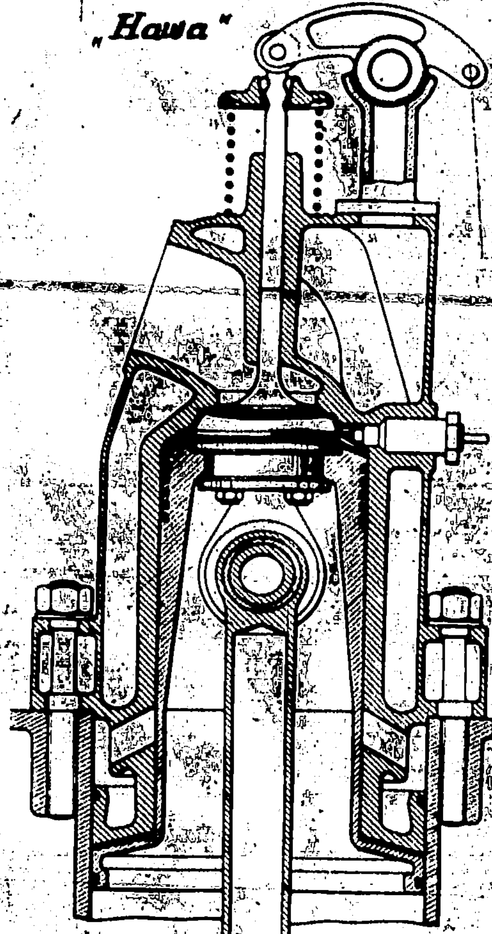

Fig.33

Huebotter

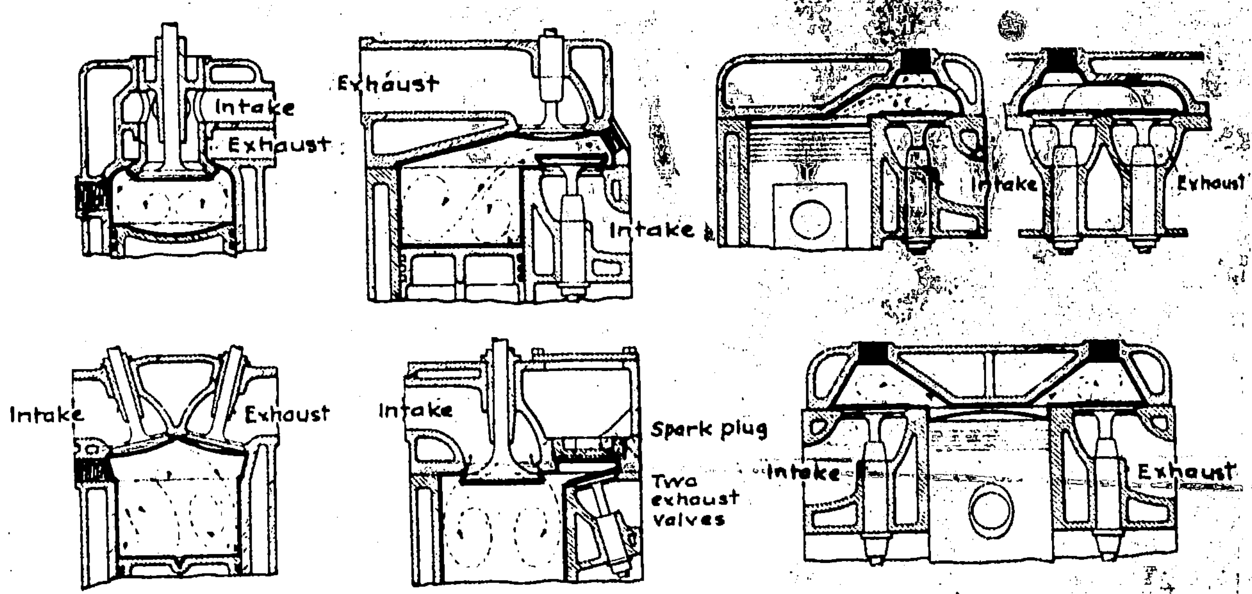

Fig.35

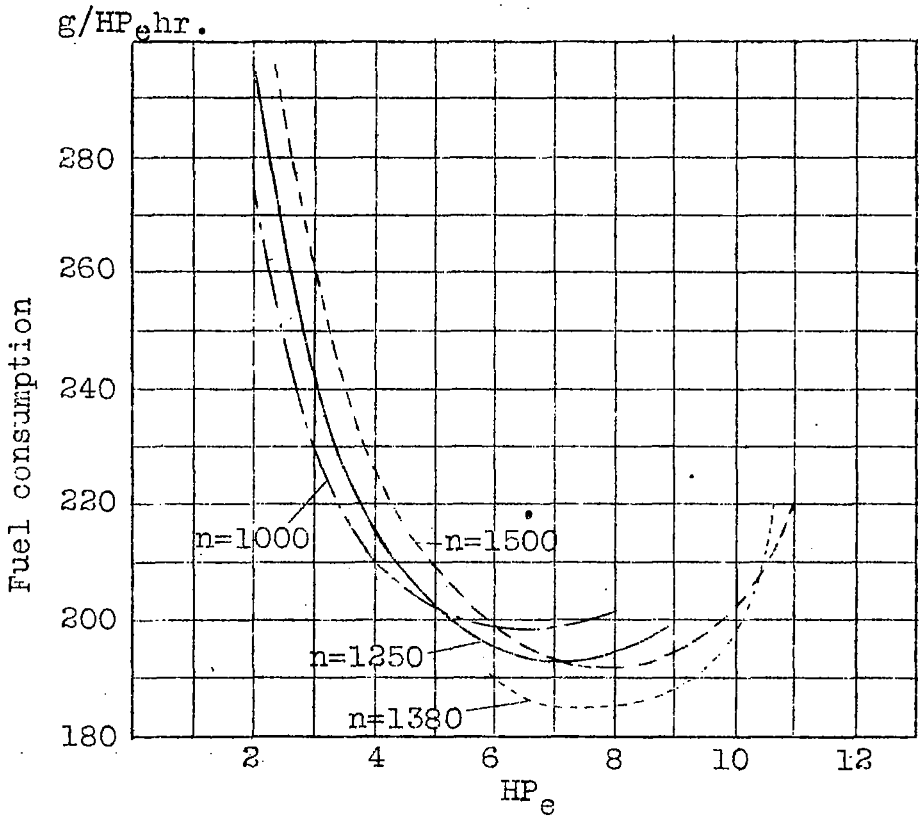

Fig.34 Acro four-stroke-cycle engine.

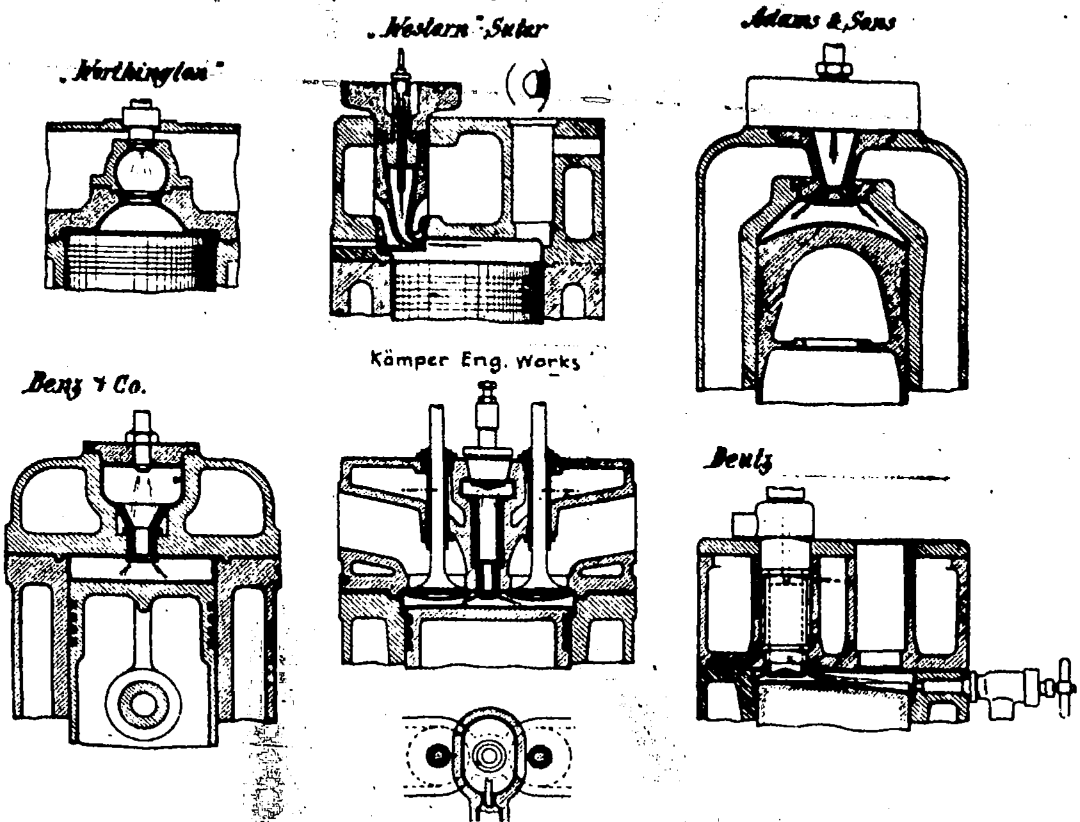
"Worthington"
"Western"-Suter
Adams & Sons
Benz & Co.
Kämper Eng. Works
Deutz

Fig.36

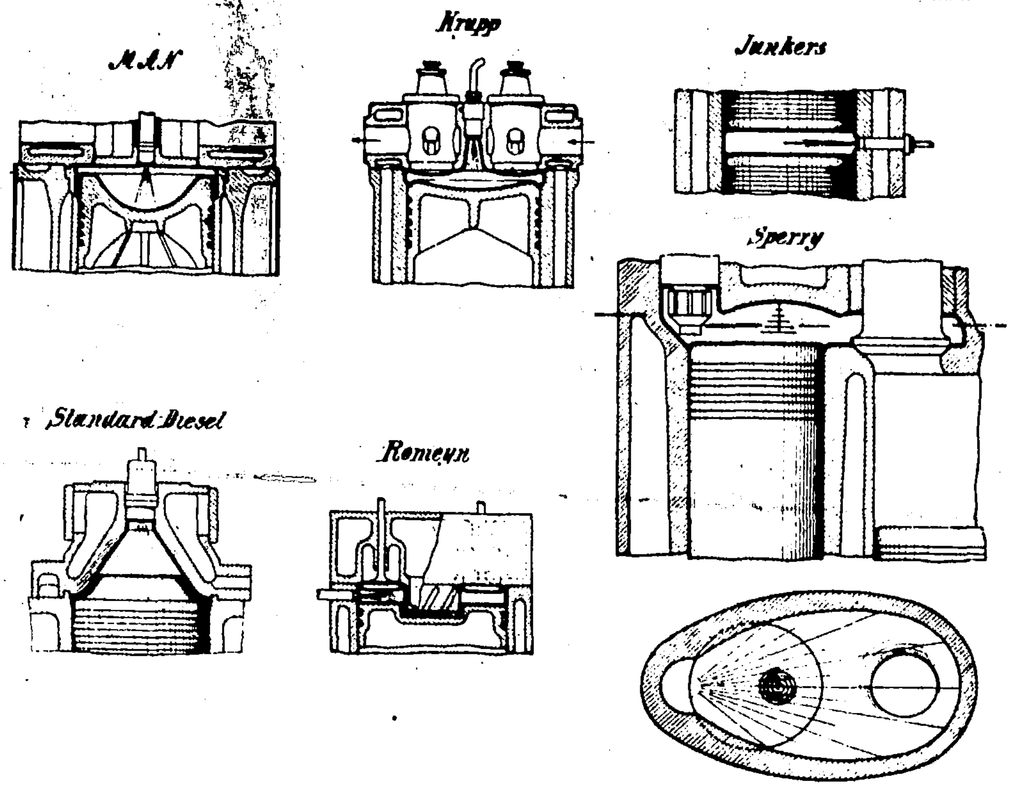
Krupp
M.A.N
Junkers
Sperry
Standard Diesel
Romeyn

Fig.37